EFFECTS OF ANOMALIES IN HYDRAULIC ACTUATORS INTO DYNAMICS OF THE AIRCRAFT CONTROL SYSTEMS

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The hydraulic servo-actuators are the important elements of the aircraft control systems. The anomalies in the system parameters of the hydraulic servo actuators cause the changes in the dynamics of the control systems having influences on the aircraft maneuverability and flight safety. The paper gives some results of practical and theoretical investigation of the hydraulic servo-actuators with mechanical feedback. The measurement of the actuator had been organized on the natural system of fighter MiG-21 that was equipped with a special load system designed for this aim and modeled the real load. The theoretical investigation deals with the determination of possibility of using the linear model of hydraulic servo-actuator for study the influences of system parameter uncertainties and anomalies on the system characteristics. There were used the estimated linearized and nonlinear models of hydraulic servo-actuators. The effects of system parameter uncertainties and nonlinearities in the hydraulic servo-actuator into the dynamics of aircraft control system were investigated in details. Here will be presented several figures with results getting by simulation and application of methods of sensitivity theory.

Nomenclature

A - state matrix,
A_p- piston area (mm^2),
B - control matrix,
E - bulk modulus (bar),
k_0 - measured and real input signals ratio (-),
k_C - Coulomb friction coefficient (N),
k_f - friction coefficient (kg/s),
k_l - loading coefficient (kg/s^2),
k_p - back flow coefficient (mm^3/bar/s),
k_x - $Q = f(x)$, flow rate coefficient (mm^3/s),
k_v - viscous friction coefficient (kg/s),
l_1, l_2 - the lengths of the rods in the feedback (mm),
m - mass of the piston with the moved elements (kg),
p_1 - pressure in the inlet to cylinder (bar),
p_2 - pressure in the outlet to cylinder (bar),
p_i - pressure in the inlet flow, supply pressure (bar),
p_o - pressure in the outlet from the servo actuator (bar),
Q_1 - inlet flow rate (mm^3/s),
Q_2 - outlet flow rate (mm^3/s),
u - input or control vector,
V_0- volume of the cylinder (mm^3),
x - control rod displacement (mm),
x - state vector,
y -- output or measurable vector,

Introduction

Nowadays the dynamics, the accuracy of hydraulic servo-actuators (HSA) play the one of the most important role in the synthesis of the several systems like direct lift control, control of the super-maneuverability aircraft, etc.. Therefore the investigation of the internal dynamic processes of the servo-actuator elements and the study the influences of system peculiarities and anomalies on the system characteristics are the actual theoretical and practical tasks.

The hydraulic servo-actuator has some nonlinearities. May be most interesting and complicated of them is the Coulomb friction of piston that can not be lienarized by simple way. So, there is a very interesting problem, how can be used the linearized model of hydraulic servo-actuator for investigation of the system anomalies.

The goal of this paper is giving the some results of practical and theoretical investigation of the hydraulic servo-actuator with mechanical feedback. The measurement of the actuator had been organized on the natural system of fighter MiG-21 that was equipped with a special load system designed for this aim and modeled the real load. The theoretical investigation deals with the determination of possibility of using the linear model of hydraulic servo-actuator for study the influences of system parameter uncertainties and anomalies on the system characteristics

1. Mathematical model of the hydraulic servo actuator

The principal schema of the investigated hydraulic servo actuator is shown on the figure 1. This is the linear symmetric 4-way servo actuator with the mechanical feedback.

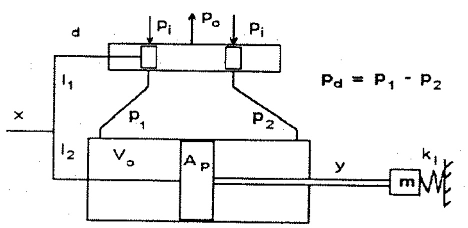


Fig. 1. The simplified model of the investigated hydraulic servo actuator

The dynamics of the HSA can be investigated on the basis of the dynamics of hydraulic and mechanical part. In first case the balance of the inlet and outlet flow rate should be written.

$$\dot{y} A_p = Q_1 - Q_2 \tag{1}$$

The flow rate depends on the input control signal of the servo actuator, eg. from the displacement, x of the control rod. The relationship of the $Q = f(x)$ has a considerable nonlinearities shown on the figure 2. This function can be linearized by using the flow rate coefficient k_x. With taking into account the effects of mechanical feedback, the resulting flow rate can be given by the following formula:

$$Q = -k_x \frac{L_1}{L_2} + k_x \frac{L_1 + L_2}{k_o L_2} x \quad . \tag{2}$$

The resulting flow rate is modified by the flow passing back in the hole between the piston and the wall of the cylinder and the compressibility of the hydraulic oil. The flow back can be taken into account by simplified linear model as the function of the different in the pressure, p_d, before and after the piston.

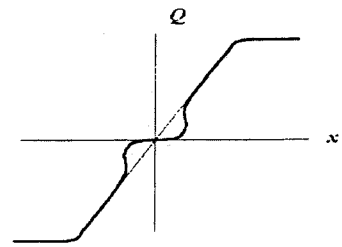


Fig. 2. Nonlinear relationship between the resulting flow rate $Q = Q_1 - Q_2$ and control rod displacement (simplified model)

The compressibility of the oil has a significant influence on the dynamics of the hydraulic actuator. The oil compressed in a cylinder compartment act like a spring and introduces therefore the second order mass-spring system whose natural frequency limits the bandwidth of any hydraulic servo abruptly[1]. The compressibility of hydraulic oil, trapped in a volume V under pressure p is defined by

$$\frac{\Delta V}{V} = \frac{\Delta p}{E} \quad . \tag{3}$$

The bulk modulus may be as high as 15 000 bar, however, in practice, is much lower, mainly due to a small percentage of entrained air, which in fact can not be avoided. According to our investigation the bulk modulus equals to 4 000 - 5 000 bar.

Finally the flow rate balance can be written in the form

$$\frac{V_o}{2E}\dot{p}_d(x) = -k_p(x) p_d(x) - A_p \dot{y} - k_x(x)\frac{L_1}{L_2} + k_x(x)\frac{L_1 + L_2}{k_o L_2} x \quad . \tag{4}$$

The second equation is based on the Newton's law. The forces should be taken into account are listed below:

- oil pressures p_1 and p_2 causing opposing forces $p_1 A_p$ and $p_2 A_p$, what can be given in the general form, $p_d A_p$, but through the pressure it is having the considerable nonlinearity shown on the figure 3.

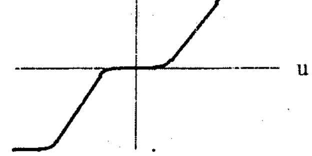


Fig. 3 Change in pressure depending on the input signal (simplified model)

- external loading force coming from outside of hydraulic actuator and as usually depends on the displacement of the hydraulic actuator. in our case. it depends on the aerodynamical and flight mechanical characteristics. too. and can be described as the "loading spring" with using the spring coefficient, k_l.
- friction force having two different parts, the Coulomb and viscous friction forces as it is shown on the figure 4.

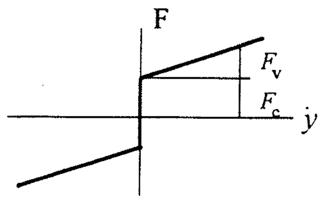


Fig. 4. The internal friction forces in the servo actuator

The Coulomb friction is the most magic non-linearising factor in hydraulic servo's and is always suspect when something irregular is happening. In normal practice, Coulomb friction can be 5 - 15 % of the maximum load-capacity. Principally it can not be neglected[1].

The viscous friction is proportional to the speed $\dot{y}$ and contributes the stability and damping in many cases.

In first approaching, the second equation can be written in the following form:

$$m\ddot{y} = p_d(x)A_p - k_C\,\mathrm{sign}(\dot{y}) - k_x(x)\dot{y} - k_l(y)y. \quad (5)$$

Generally, the equations (4) and (5) easy can be rewritten in the linearized form:

$$\frac{V_o}{2E}\dot{p}_d = -k_p p_d - A_p\dot{y} - k_x\frac{L_1}{L_2}y + k_x\frac{L_1+L_2}{k_o L_2}x,$$
$$m\ddot{y} = p_d A_p - k_f\dot{y} - k_l y. \quad (6)$$

Here the coefficients, k, are the constants. The k_f is the friction coefficient taking into account the effect of Coulomb and viscous friction forces.

2. Practical investigation of the hydraulic servo actuators

The practical investigation of the real servo actuator was organized at the Szolnok College of Flight Officers. Real hydraulic system of fighter MiG-21 set up in laboratory was equipped by a special spring load system designed for this investigation and modeled the real load situations. The displacements of the control and the piston rods were measured by the Hottinger sensors of linear distances. The inlet and outlet pressures were measured at the connection of the system to the servo actuator by the Hottinger pressure sensors. In principle the measured pressures (Fig. 5.) are not those pressures what are used in the models, but the different in the inlet and outlet pressure was calculated quit easy from the measured data.

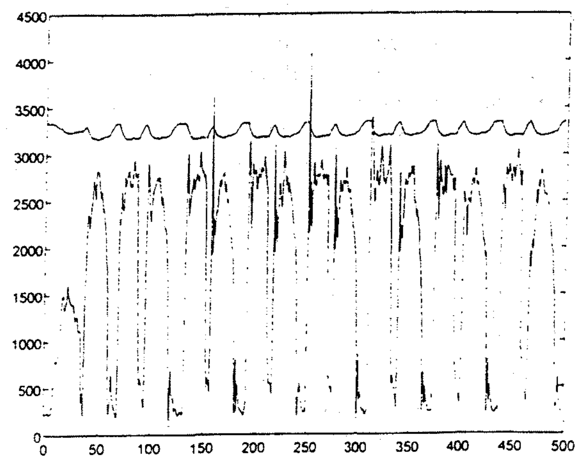


Fig. 5. The example of the measured inlet and outlet pressures in form of storage in computer memory

The accoracy of sensors was less than $\pm$ 1 %. The measurement was controlled by computer and the data were collected in memory of the computer. The sample frequency of measurement was changed from 50 to 7500 Hz. We found that, the changes in pressure generated by the pressure waves can be measured safety in case of frequency 3000 Hz at least. But the dynamics of servo actuator already can be checked reliable in measuring with 30 - 50 Hz sampling frequency. In the measurement the 100 Hz sampling frequency and 10 bit A/D converter were applied.

The measurements were done with different input functions acting to the control rod. The effects of failures were investigated in cases when the system elements were changed. The collected data after preliminary data processing (fig. 6.) were used for identification of the model of the hydraulic servo actuator.

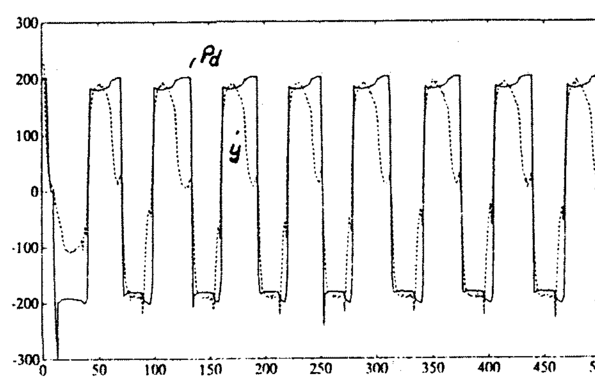


Fig. 6. The typical measurement result

3. Identification of the simplified model

The linearized model of the hydraulic servo actuator (6) can be rewritten in form of state space representation

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u}$$

$$\mathbf{x} = \begin{bmatrix} p_d = \text{different in the inlet and outlet pressure} \\ \dot{y} = \text{acceleration of piston} \\ y = \text{displacement of piston} \end{bmatrix} \quad (7)$$

$$\mathbf{u} = \left[x = \text{displacement of control rod} \right]$$

The MATLAB Identification toolbox had been used for identification of the prescribed model. Unfortunately, this software did not give the acceptable results, because the state matrix is too close to the unstable matrix. Therefore the specially developed identification softwares based on the least square methods and random minimization of the performance index. Finally the identified linear model had the following values:

$$\mathbf{A} = \begin{bmatrix} -0.0052 & -41.284 & -949.56 \\ 3.27*10^6 & -3.65*10^6 & 1.406*10^3 \\ 0 & 1 & 0 \end{bmatrix}, \qquad (8)$$

$$\mathbf{B} = \begin{bmatrix} 847.4 \\ 0 \\ 0 \end{bmatrix}.$$

The different in the identified linear model and real object is demonstrated on the figures 7 - 9.

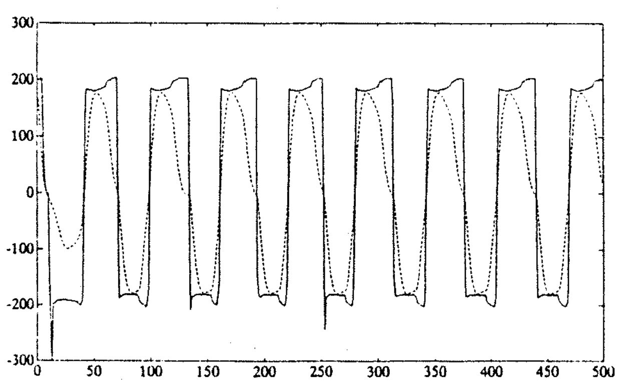


Fig. 7. The measured (—) and simulated (---) by application of the identified linear model changes in the different between the inlet and outlet pressures

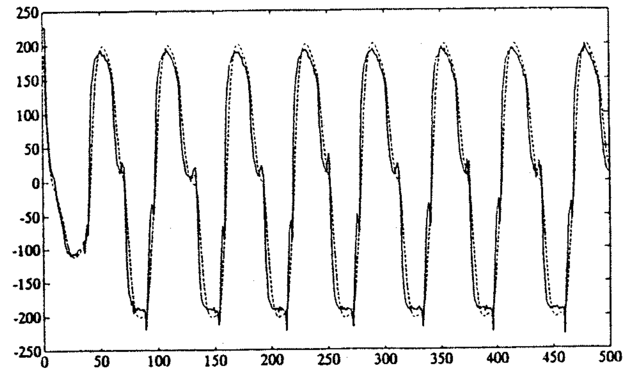


Fig. 8. The measured (—) and simulated (---) acceleration of the piston of hydraulic servo actuator

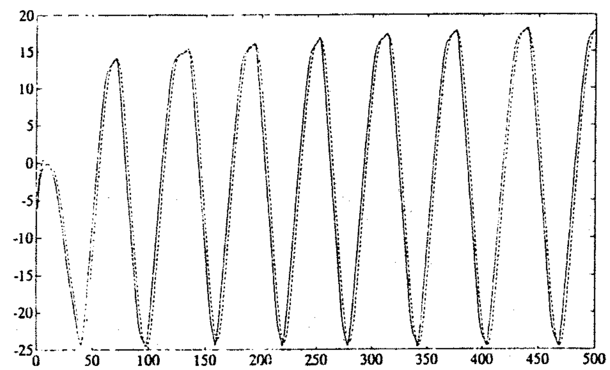


Fig. 9. The measured (—) and simulated (---) motion of piston of the servo actuator

4. The theoretical investigation

After large practical investigation the effects of nonlinearities and anomalies e.g. small deviations in the characteristics on the dynamics of hydraulic servo actuator were investigated theoretically. We found that, the general model completed from the equations (4) and (5) can be integrated by application of the 4 order Runge Kutta method with steps less than 10^{-7}. The nonlinearities discussed in the chapter 1 have influence on the dynamics only in case of the input changing with high frequency (Fig.10).

On the basis of practice gained in the practical investigation and data processing, the real hydraulic servo actuator model had been developed and investigated with connection with the aircraft longitudinal motion model[2].

The models were defined in the state space representation with the following state and control vectors:

servo actuator

$\mathbf{x} = [\ \varphi,\ d\varphi/dt,\ p_d\ ]^T$, $\mathbf{u} = [\ 0,\ 0,\ x]^T$,

aircraft motion

$\mathbf{x} = [\ \Delta\alpha,\ \Delta v,\ d\Delta v/dt\ ]^T$, $\mathbf{u} = [\ 0,\ 0,\ \varphi\ ]^T$.

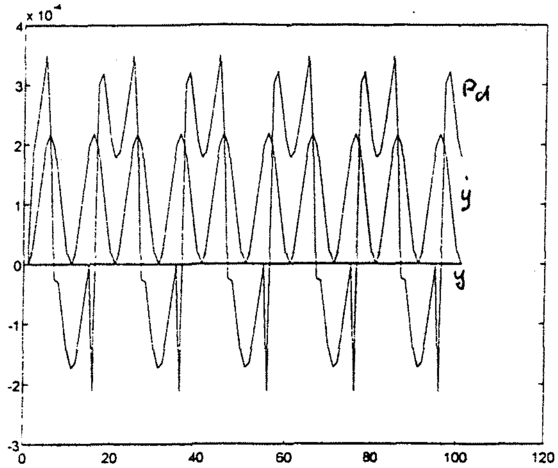
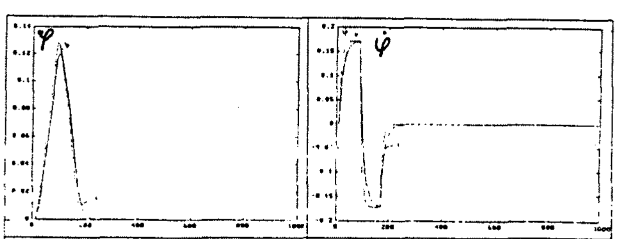


Fig. 10. The highly nonlinear effects on the output signals of the hydraulic servo actuator in case of the sinusoidal input with frequency 100 Hz.

The effects of deviation in the state matrix element were investigated through the integration of the system and determining the sensitivity functions. The theoretical investigations are shown that, the small deviations in the parameters of the servo actuator do not have considerable effects on the servo actuator and aircraft dynamics (Fig. 11 - 13)

On the Figure 11. the effect of 50 % change in the state matrix element $a_{3,2}$ of the servo actuator model on the output characteristics. The smaller changes in the state matrix elements are generate mach less, negligible effects.



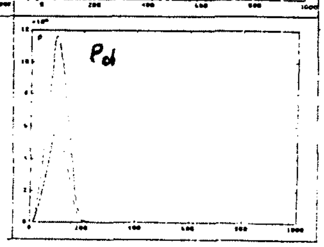
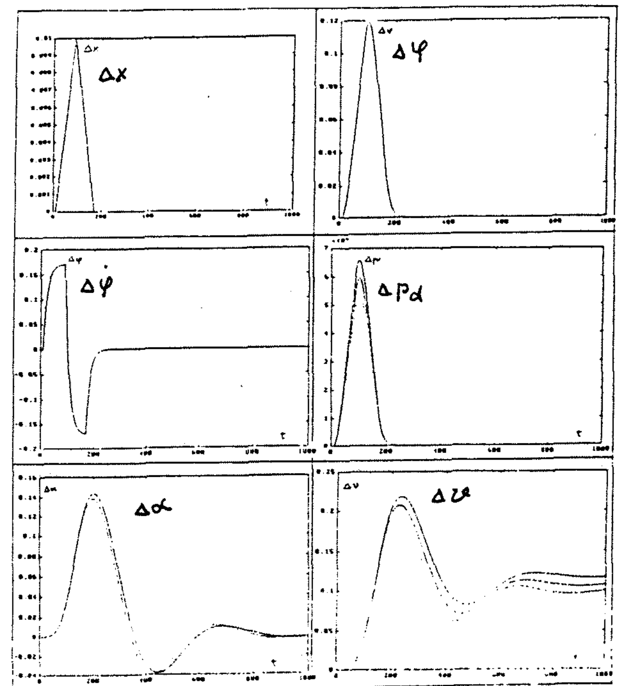


Fig. 11.

Effects of 50 % change in HSA state matrix element $a_{3,2}$ on the output characteristics

On the Figure 12 the effects of changes in the HSA state matrix elements on the flight mechanical characteristics are demonstrated. The aircraft motion model has mach more sensitivity to changes in the HSA parameters, but these effects are not a very big, too.



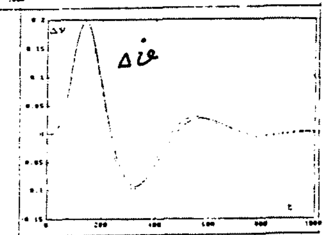


Fig. 12.

Effects of 10 % changes in HSA state matrix element $a_{3,1}$ and $a_{3,2}$ on the flight mechanical characteristics

The typical sensitivity functions are shown on the fig. 13. As it can be seen the changes in the parameter of the HSA have the very small effects on the dynamics of aircraft motion.

Conclusion

On the basis of the practical and theoretical investigation we can do the following conclusions:

- the dynamics of the hydraulic servo actuators can be investigated with sampling frequency 30 -50 Hz,
- the changes in the oil pressure should be measured sampling frequency 3 000 Hz at least,
- in case of normal application the hydraulic servo actuators have the "low dynamic" processes

- the deviations in the parameters of hydraulic servo actuators cause the small effects on the system dynamics
- the mean nonlinearities in the system dynamics of hydraulic servo actuators are the Coulomb friction and flow volume rate in the pistons,
- the nonlinearities and anomalies in the hydraulic servo actuators have the small influences on the aircraft dynamics, which should be taken into account only in case of high frequency excitation with control input.

References

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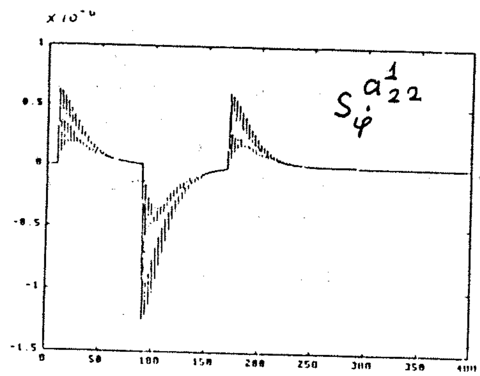

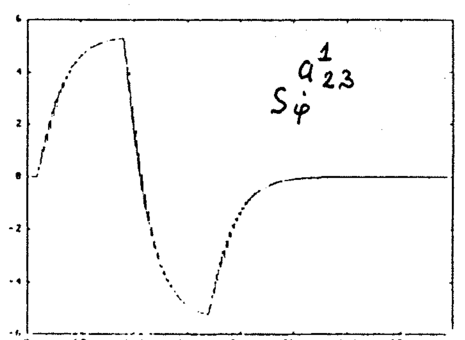

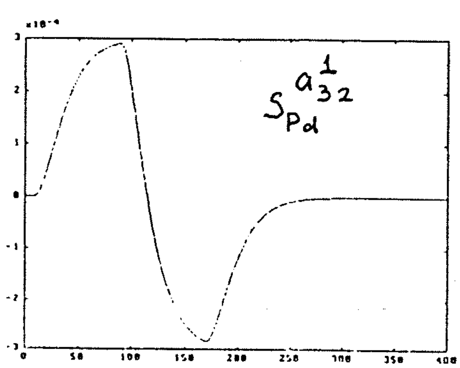

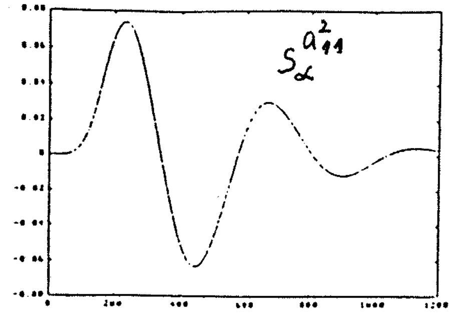

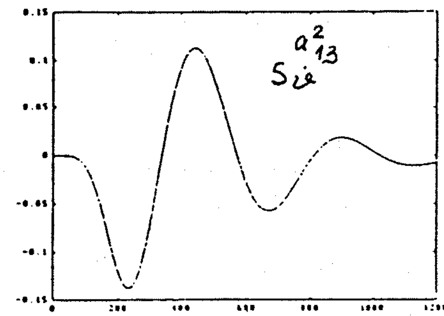

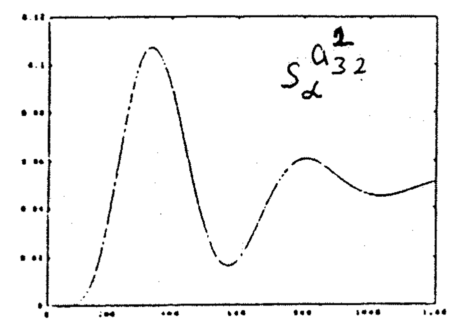

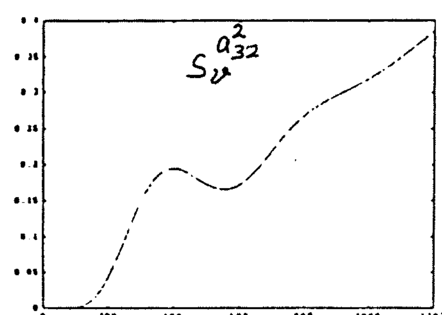

Fig. 13. The typical sensitivity functions $S_{y_k}^{a^o_{i,j}}$. (Sensitivity of y_k characteristics to the state matrix element, $a^o_{i,j}$, where o = 1 - HSA state matrix, o = 2 - aircraft state matrix, i = row, j - column)